See the chapter on Digital Analysis in Volume II for more information on these analysis techniques. The frequency domain can be divided into three main areas:

1. Synchronous components
2. Sub-synchronous components
3. Non-synchronous components

1.5.1 Synchronous Components

Synchronous components (also called **phase locked** components) occur at frequencies equal to N*RPM, where N is an integer and RPM is the rotating frequency of the shaft at the position where the measurement is taken. The **fundamental frequency** (also called the **first order**) is equal to the primary rotating speed of the shaft at the location being measured, and is equal to 1*RPM.

There are a variety of possible synchronous signals within a vibration signature:

Low multiples (where N = 1 to 8) may be caused by

- **imbalance**: occurs at 1*RPM. If there is looseness within the system then there may be several harmonics of the first order.
- **pitch line runout on belt sheaves**: causes a vibration at 1*RPM and is often confused with imbalance.
- **shaft center line and bearing misalignment**: can show up at 1 or more times RPM.
- **bent shaft**: 1 or more times RPM.
- **looseness**: has many harmonics of 1*RPM frequency; second or third order harmonics often have a higher amplitude than the first order. If the problem is pure looseness, then the harmonics are mostly odd-order (3*RPM, 5*RPM, etc.).
- **blade or vane pass** (small number of blades or vanes): occurs at number of blades or vanes times RPM.

- **cavitation**: the implosion of a void on the intake blade of a fan or pump when the void reaches the pressure side. Cavitation occurs at 1*RPM with harmonics up to the number of blades. High frequency noise is also present.
- **non-linear motion**: a reciprocating or geneva motion causes vibration at multiples of the first order. In most cases the second order is the highest signal.
- **non-linear response**: harmonics of operating frequency can be caused by non-linearities in the mechanical system such as cracks or other discontinuities, non-linear damping, or non-linear spring constants.

High multiples (where N > 8) may be caused by

- **gear mesh frequencies**: occurs at the number of gear teeth * RPM. These frequencies can be very high; frequencies of 8,000 to 15,000 Hz are not unusual.
- **blade pass** (on compressors or turbines): occurs at the number of blades * RPM.
- **slot frequency on motors**: the number of rotor slots on most electric motors is greater than 30, creating a high slot pass frequency.

1.5.2 Sub-Synchronous Components

Sub-synchronous components are frequencies that are less than 1*RPM of the shaft in the bearing where the data was taken. These have many sources:

- another component within the machine.
- another machine forcing this machine to vibrate.
- **primary belt frequency**: can be calculated by using the formulas shown in Section 1.3.5. Higher harmonics may also exist.
- **oil whirl and oil whip**: occurs at approximately 40 to 48% of 1*RPM. See Section 1.3.3 for a discussion of oil whirl and oil whip.
- **rubs**: occurs very close to 1/3 or 1/2 of 1*RPM, and higher 1/3 or 1/2 order harmonics are usually present.
- **a loose anti-friction bearing in its housing**: occurs at 50% of 1*RPM, 1.5*RPM, 2.5*RPM etc.
- **FTF of ball bearings**: varies from bearing to bearing and, if present, usually indicates advanced stage of bearing failure.
- **a surge**: usually occurs from 10 to 50% of 1*RPM.

1.5.3 Non-Synchronous Components

Non-synchronous components are frequencies that are higher than 1*RPM but are not synchronous to the frequency of the shaft in the bearing where the data was taken. However, harmonic families of some forcing functions which are *not* at the rotational frequency of the shaft may also be present. A typical example is a bearing fault frequency, where first, second, and third order vibrations of an inner-race or outer-race fault are usually present.

Typical non-synchronous components include

- another component in the machine.
- another machine in the area.

- other system resonances.
- belts: in most cases the highest belt frequency is equal to the primary belt frequency times the number of pulleys. In multiple-belt systems, frequencies can occur at 2*primary belt frequency up to twice the number of belts times the primary belt frequency.
- bearing fault frequencies: since the outer race of a bearing is always under load, the outer race fault frequency is most common. Outer race frequency is approximately equal to the number of balls times FTF (the fundamental cage frequency, which is approximated by 0.4*RPM).
- electrical: electrically-caused vibrations include loose transformer laminations, the dress of conductors in a raceway, electric motors with broken or cracked rotor bars, eccentric rotors or stators, rotors loose on the motor shaft, motor rotor not magnetically centered, etc.

The background vibration levels of the machine should also be checked when the machine is shut down. In more than one case extensive analysis has tried to identify a vibration problem in a machine, when the vibration was present even when the machine was shut off!

If background vibration (caused by other machines) is repetitive, the background levels can be removed from the frequency spectrum by negative averaging.

TIME DOMAIN ANALYSIS		
TYPE OF WAVEFORM	**PROBABLE CAUSE**	**COMMENTS**
Impacts	Looseness	
	Bearing fault frequency	
	Cracked shaft	
	Fan wheel loose on shaft	
Non-periodic components	Foreign material	
	Outside source of vibration	
	Surge	
Asymmetrical waveforms	Rubs	
	Restrictions to motion	

0.15 = 3.81, 0.12 = 3.04, 0.1 = 2.54, 0.18 = 4.57, 0.08 = 2.03

FREQUENCY DOMAIN ANALYSIS

FREQUENCY	HARMONICS	AMPLITUDES	PROBABLE CAUSE	COMMENTS
1 * RPM	No	1*RPM H > .15 in/sec 1*RPM V > .12 in/sec Low axial both bearings	Imbalance	Consistent 1*RPM phase
		1*RPM H > .15 in/sec 1*RPM V > .12 in/sec 1*RPM A > .1 in/sec	Bent shaft	Look for axial phase 180° out
		1*RPM H > .18 in/sec 1*RPM V < .08 in/sec	Horizontal resonance	High amplitude may be at some other order excited by non-linear system
		1*RPM H < .08 in/sec 1*RPM V > .18 in/sec	Vertical resonance	
		Increases after fan startup	Thermal bow	
		High motor 1*RPM at fan bearing	BELT DRIVE: - Pitch line runout on motor pulley - Fan mount resonance at motor speed	
		High fan 1*RPM at motor bearing	BELT DRIVE: - Pitch line runout on fan pulley - Motor mount resonance at fan speed	
	2*RPM	1*RPM H > .18 in/sec 1*RPM V < .08 in/sec	Horizontal resonance plus vertical misalignment	Phase across direct drive coupling will either be in phase or ± 10° out of phase
		1*RPM H < .08 in/sec 1*RPM V > .18 in/sec	Vertical resonance plus horizontal misalignment	
			Disk skew	Vibration builds as in resonance but phase does not change
	2*RPM, 3*RPM	1*RPM H > .15 in/sec 1*RPM V > .12 in/sec Low axial both bearings	Imbalance plus system non-linearity (looseness, cracks, excessive clearance)	- Consistent 1*RPM phase - Can also produce ½ order harmonics
		High axial on bearing	Bearing cocked on shaft Both bearings are fixed (rare but possible)	
	Up to number of blades x RPM	H, V, or A vibration > .1 in/sec	AXIAL FAN: non-uniform blade track	
			AXIAL FAN: - Blade pitch error - Structural element too close to fan blades	Higher harmonics may also be present

0.15 = 3.81, 0.12 = 3.04, 0.1 = 2.54, 0.18 = 4.57, 0.08 = 2.03

FREQUENCY DOMAIN ANALYSIS

FREQUENCY	HARMONICS	AMPLITUDES	PROBABLE CAUSE	COMMENTS
1*RPM	3*RPM, 4*RPM ½ orders		- Locked coupling - External side force	- Locked coupling is usually load-dependent - Asymmetric time signal may also be present
	2*RPM-4*RPM higher than 1*RPM		- Bearings cocked on shaft - Bearings not in same plane	Axial vibration on the bearings will be high
			Cavitation	- High frequency noise also present - Wide skirts
	Group at belt frequency		Belt misalignment	
# blades * RPM	No	H, V, A vibration > .15 in/sec	- Fan operating too far from best efficiency point - Fan cutoff too close to blades - Structural resonance AXIAL FAN: non-uniform blade pitch	
Belt frequency	# pulleys order usually highest		- loose belts - faulty belts	Harmonics can be as high as 9th or 10th order
# belts * belt frequency			Shafts not parallel	
Belt strand resonance			Resonance excited by belt strand	Wide skirt, since resonance of tight and loose strands are within 3-6 Hz of each other
Approximately 40% of 1*RPM	No		Oil whirl	Appears as inside loop on shaft orbit
33-50% of 1*RPM	No		Fan in surge	- Accompanied by pulsations in duct - High noise usually present
60-75% of 1*RPM	Yes		Rotating stall	- caused by inlet disturbance - check damper design and installation

SUMMARY

Fans are designed to move air against moderate pressures (the total air resistance in a duct system). The pressure, temperature, density, viscosity, and humidity of the air all affect the design and performance of a fan.

There are two primary types of fans: centrifugal and axial flow.

Centrifugal fans consist of an impeller running in a spirally-shaped casing. Usually the air enters the casing from the impeller's axial direction and is discharged at the periphery of the impeller.

The flow of air is parallel to the impeller axis in an axial flow fan. Axial fan blades are usually an airfoil in cross-section, and run at a very close clearance to the side of the fan casing. Air enters the fan in an axial direction. The air leaving an axial flow fan has both an axial and a rotational component.

5.0 ANALYSIS

Refer to the following tables to determine the probable cause of pump vibration.

TIME DOMAIN ANALYSIS		
TYPE OF WAVEFORM	**PROBABLE CAUSE**	**COMMENTS**
Impacts	Looseness	
	Bearing fault frequency	
	Cracked shaft	

0.15 = 3.81, 0.12 = 3.04, 0.1 = 2.54, 0.18 = 4.57, 0.08 = 2.03

TIME DOMAIN ANALYSIS		
TYPE OF WAVEFORM	PROBABLE CAUSE	COMMENTS
Non-periodic components	Foreign material	
	Outside source of vibration	
Asymmetrical waveforms	Rubs	
	Restrictions to motion	

FREQUENCY DOMAIN ANALYSIS				
FREQUENCY	HARMONICS	AMPLITUDES	PROBABLE CAUSE	COMMENTS
1 * RPM	No	1*RPM H > .15 in/sec 1*RPM V > .12 in/sec Low axial both bearings	Imbalance	Consistent 1*RPM phase
		1*RPM H > .15 in/sec 1*RPM V > .12 in/sec 1*RPM A > .1 in/sec	Bent shaft	Look for axial phase 180° out
		1*RPM H > .18 in/sec 1*RPM V < .08 in/sec	Horizontal resonance	High amplitude may be at some other order excited by non-linear system
		1*RPM H < .08 in/sec 1*RPM V > .18 in/sec	Vertical resonance	
		Increases after startup	Thermal bow	
		High motor 1*RPM at pump bearing	Pump mount resonance at motor speed	
		High pump 1*RPM at motor bearing	Motor mount resonance at pump speed	
		High axial	Hydraulic imbalance	- Cannot be corrected by mechanical balancing - Usually caused by poorly designed suction piping
		High radial	Imbalanced impeller	
	2*RPM	1*RPM H > .18 in/sec 1*RPM V < .08 in/sec	Horizontal resonance plus vertical misalignment	Phase across direct drive coupling will either be in phase or ± 10° out of phase
		1*RPM H < .08 in/sec 1*RPM V > .18 in/sec	Vertical resonance plus horizontal misalignment	
	2*RPM, 3*RPM	1*RPM H > .15 in/sec 1*RPM V > .12 in/sec Low axial both bearings	Imbalance plus system non-linearity (looseness, cracks, excessive clearance)	- Consistent 1*RPM phase - Can also produce ½ order harmonics
		High axial on bearing	Bearing cocked on shaft	
	3*RPM, 4*RPM ½ orders		- Locked coupling - External side force	- Locked coupling is usually load-dependent - Asymmetric time signal may also be present

0.15 = 3.81, 0.12 = 3.04, 0.1 = 2.54, 0.18 = 4.57, 0.08 = 2.03

FREQUENCY DOMAIN ANALYSIS				
FREQUENCY	**HARMONICS**	**AMPLITUDES**	**PROBABLE CAUSE**	**COMMENTS**
1*RPM	2*RPM-4*RPM higher than 1*RPM		- Bearings cocked on shaft - Bearings not in same plane	Axial vibration on the bearings will be high
			Cavitation	- High frequency noise also present - Wide skirts
	Sub-harmonics from 5% to 35% of operating frequency		Recirculation	Pulsating and/or loud cracking noise in suction piping
# blades x RPM	No	H, V, A vibration > .15 in/sec	Acoustic resonance	May be caused by interaction of impeller vanes with volute or diffusion vanes
Approximately 40% of 1*RPM	No		Oil whirl	Appears as inside loop on shaft orbit
5-45% of 1*RPM	No	High level, random	- Volute/diffuser vane interaction - Excessive clearance between impeller and casing	Usually caused by operation away from best efficiency point
50% of 1*RPM	½ orders		- Rubs in wear ring area - Anti-friction bearings loose in housing	

SUMMARY

Pumps move fluid from one point to another by adding energy (not pressure) to the fluid. Since energy is defined as the ability to do work, adding energy to a fluid causes the fluid to do work: move through a pipe, fill a tank, or flow to a higher level.

The mechanical energy that is imparted to the fluid is independent of the fluid's density and is called the **pump head**. Pump head is measured in foot-pounds per pound of fluid.

Pump head is measured in feet, and the head generated by a pump operating at a given speed and capacity will be the same for all fluids. Therefore pump heads are expressed in feet of liquid. Head is the height of the fluid, while pressure is the force exerted by the fluid by virtue of its height.

The total head against which the pump must operate is called the **system head**. System head is made up of static head, friction head, and velocity head. The available head (total suction head - vapor pressure in feet) is called the "net positive suction head," and is abbreviated **NPSH**.

The velocity of the fluid must decrease when the tube's cross-section increases. Likewise, the velocity increases when the tube's cross-section decreases. The sum of a fluid's pressure, density, and velocity head are equal at any point along the flow of an ideal fluid. Therefore if a fluid's velocity increases, then the pressure drops; likewise, the pressure increases as the velocity decreases.

There are two main types of pumps:

1. Centrifugal pumps
2. Axial pumps

6.0 ANALYSIS

The following tables can help you determine the probable cause (or causes) of the vibration you measured.

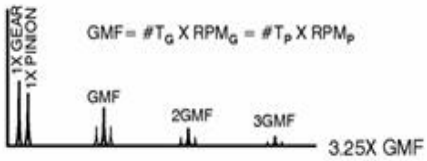
TIME DOMAIN ANALYSIS		
TYPE OF WAVEFORM	**PROBABLE CAUSE**	**COMMENTS**
Impacts	Looseness	
	Bearing fault frequency	
	Cracked shaft	
	Rotor loose on shaft	
Non-periodic components	Foreign material	
	Outside source of vibration	
Asymmetrical waveforms	Rubs	
	Restrictions to motion	

0.15 = 3.81, 0.12 = 3.04, 0.1 = 2.54, 0.18 = 4.57, 0.08 = 2.03

FREQUENCY DOMAIN ANALYSIS				
FREQUENCY	**HARMONICS**	**AMPLITUDES**	**PROBABLE CAUSE**	**COMMENTS**
1 * RPM	No	1*RPM H > .15 in/sec 1*RPM V > .12 in/sec Low axial both bearings	Imbalance	Consistent 1*RPM phase
		1*RPM H > .15 in/sec 1*RPM V > .12 in/sec 1*RPM A > .1 in/sec	Bent shaft	Look for axial phase 180° out
		1*RPM H > .18 in/sec 1*RPM V < .08 in/sec	Horizontal resonance	High amplitude may be at some other order excited by non-linear system
		1*RPM H < .08 in/sec 1*RPM V > .18 in/sec	Vertical resonance	
		Increases after startup	Thermal bow	
	No	High motor 1*RPM at bearing	Motor mount resonance at motor speed	
	2*RPM	1*RPM H > .18 in/sec 1*RPM V < .08 in/sec	Horizontal resonance plus vertical misalignment	Phase across direct drive coupling is either in phase or ± 10° out of phase
		1*RPM H < .08 in/sec 1*RPM V > .18 in/sec	Vertical resonance plus horizontal misalignment	
	2*RPM, 3*RPM	1*RPM H > .15 in/sec 1*RPM V > .12 in/sec Low axial both bearings	Imbalance plus system non-linearity (looseness, cracks, excessive clearance)	- Consistent 1*RPM phase - Can also produce ½ order harmonics
		High axial on bearing	Bearing cocked on shaft	
	3*RPM, 4*RPM ½ orders		- Locked coupling - External side force	- Locked coupling is usually load-dependent - Asymmetric time signal may also be present
	2*RPM-4*RPM higher than 1*RPM		- Bearings cocked on shaft - Bearings not in same plane	Axial vibration on the bearings will be high
	Slot pass freq. with sidebands at 2*line freq.		Eccentric rotor	- Vibration will beat at freq. equal to # poles * slip freq. - Disappears when power is cut
2*slip frequency	Sidebands of line freq. at ± the # poles * slip		Broken rotor bars	Similar to eccentric rotor, but beat at # poles * slip is proportional to load
2*line frequency			- Rotor not centered in stator - Stator loose in motor frame - Loose laminations or coils in stator	- Vibration is regardless of number of poles - Vibration increases with load and disappears when power is cut
Rotor frequency * # of stator slots			Core slot pass frequency	May excite system resonance

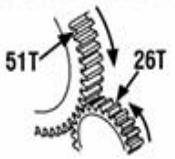
GEARS

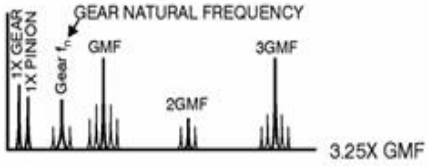
A. NORMAL SPECTRUM



Normal Spectrum shows Gear & Pinion Speeds, along with Gear Mesh Frequency (GMF) and very small GMF harmonics. GMF harmonics commonly will have running speed sidebands around them. All peaks are of low amplitude, and no natural frequencies of gears are excited. F_{MAX} recomended at 3.25X GMF (minimum) when # teeth are known. If tooth count is not known, set F_{MAX} at 200X RPM on each shaft.

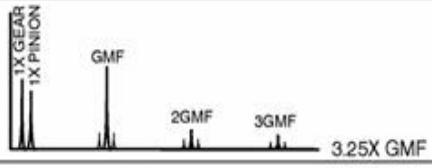
B. TOOTH WEAVE





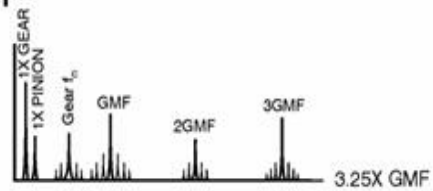
Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur when wear is noticeable. Sidebands may be better wear indicator than GMF frequencies themselves. Also, high amplitudes commonly occur at either 2XGMF or at 3XGMF (esp. 3XGMF), even when GMF amplitude is acceptable.

C. TOOTH LOAD



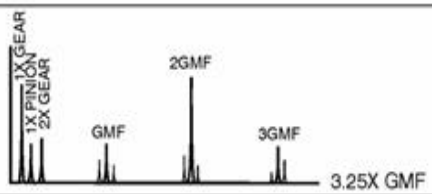
Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful spectral comparisons.

D. GEAR ECCENTRICITY AND BACKLASH



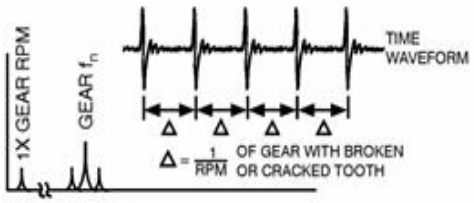
Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband frequencies.Also, 1X RPM level of eccentric gear will normally be high if eccentricity is the dominant problem. Improper backlash normally excites GMF harmonics and Gear Natural Frequency, both of which will be sidebanded at 1X RPM. GMF amplitudes will often decrease with increasing load if backlash is the problem.

E. GEAR MISALIGNMENT



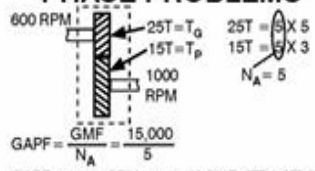
Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF will often be spaced at 2X RPM. Note that sideband amplitudes often are not equal on left and right side of GMF and GMF harmonics due to the tooth misalignment. Causes uneven wear pattern.

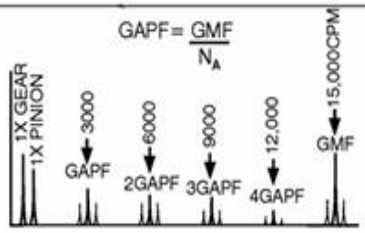
F. CRACKED/BROKEN TOOTH



A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh with teeth on the mating gear. Time between impacts (Δ) will correspond to 1/RPM of gear with the problem. Amplitudes of Impact Spikes in Time Waveform often will be 10X to 20X higher than that at 1X RPM in the FFT!

G. GEAR ASSEMBLY PHASE PROBLEMS

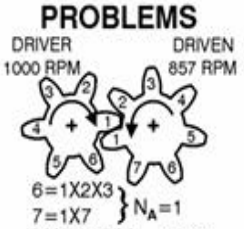


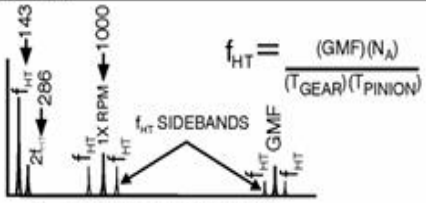


GAPF = 3000 CPM = 0.20X GMF **(FRACTIONAL GMF)**

Gear Assembly Phase Freq. (GAPF) can result in Fractional Gear Mesh Frequencies (if $N_A > 1$). It literally means (T_n/N_A) gear teeth will contact (T_n/N_A) pinion teeth and will generate N_A wear patterns, where N_A in a given tooth combination equals the product of prime factors common to the number of teeth on the gear and pinion (N_A = Assembly Phase Factor). GAPF (or harmonics) can show up right from the beginning if there were manufacturing problems. Also, its sudden appearance in a periodic survey spectrum can indicate damage if contaminate particles pass through the mesh, resulting in damage to the teeth in mesh at the time of ingestion just as they enter and leave meshing or that gears have been reoriented.

H. HUNTING TOOTH PROBLEMS



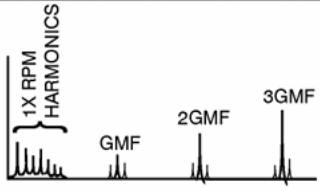


$f_{HT} = \frac{(6X\ 1000)(1)}{(6)(7)} = \frac{1000}{7}$ = 143 CPM (One Pulse Per 7 Pinion Revolutions)

Hunting Tooth Frequency (f_{HT}) occurs when faults are present on both the gear and pinion which might have occurred during the manufacturing process, due to mishandling, or in the field. It can cause quite high vibration, but since it occurs at low frequencies predominately less than 600 CPM, it is often missed. A gear set with this tooth repeat problem normally emits a "growling" sound from the drive. The maximum effect occurs when the faulty pinion and gear teeth both enter mesh at the same time (on some drives, this may occur only 1 of every 10 to 20 revolutions, depending on the f_{HT} formula). Note that T_{GEAR} and T_{PINION} refer to number of teeth on the gear and pinion, respectively. N_A is the Assembly Phase Factor defined above. Will often modulate both GMF and Gear RPM peaks.

GEARS (CONTINUED)

I. LOOSE BEARING FIT

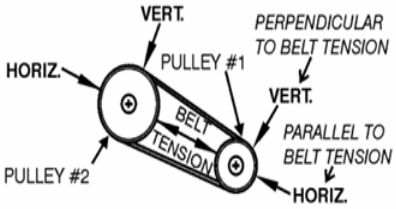


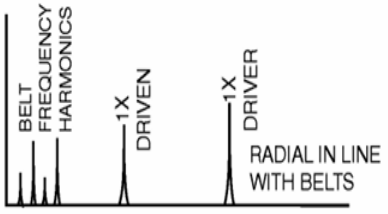
Excessive Clearance of bearings supporting the gears can not only excite many running speed harmonics, but will often cause high amplitude response at GMF, 2GMF and/or 3GMF. These high GMF amplitudes are actually a response to, and not the cause of, looseness within the bearings supporting the gearing. Such excessive clearance can be caused either by extensive bearing wear or by improper bearing fit onto the journal during installation. Left uncorrected, it can cause excessive gear wear and damage to other components.

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

BELT DRIVE PROBLEMS

A. WORN, LOOSE OR MISMATCHED BELTS





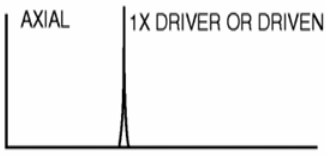
PITCH DIAM$_1$ X RPM$_1$ = PITCH DIAM$_2$ X RPM$_2$

BELT FREQ. = $\frac{3.142 \times \text{PULLEY RPM} \times \text{PITCH DIAM.}}{\text{BELT LENGTH}}$

TIMING BELT FREQ. = BELT FREQ. X #BELT TEETH
= PULLEY RPM X #PULLEY TEETH

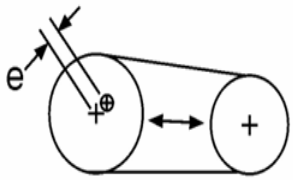
Belt frequencies are below the RPM of either the motor or the driven machine. When they are worn, loose or mismatched, they normally cause 3 to 4 multiples of belt frequency. Often 2X belt freq. is the dominant peak. Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM. On timing belt drives, wear or pulley misalignment is indicated by high amplitudes at the Timing Belt Frequency. Chain drives will indicate problems at Chain Pass Frequency which equals #Sprocket Teeth X RPM.

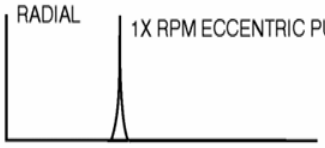
B. BELT/PULLEY MISALIGNMENT



Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor will be at fan RPM, or vice versa. Can be confirmed by phase measurements by setting Phase Filter at RPM of pulley with highest axial amplitude; then compare phase at this particular frequency on each rotor in the axial direction.

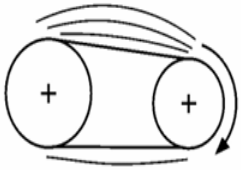
C. ECCENTRIC PULLEYS

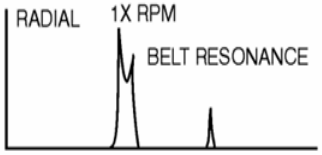




Eccentric pulleys cause high vibration at 1X RPM of the eccentric pulley. The amplitude is normally highest in line with the belts, and should show up on both driver and driven bearings. It is sometimes possible to balance eccentric pulleys by attaching washers to taper-lock bolts. However, even if balanced, the eccentricity will still induce vibration and reversible fatigue stresses in the belt. Pulley eccentricity can be confirmed by phase analysis showing horizontal & vertical phase differences of nearly 0° or 180°.

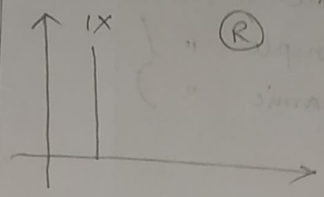
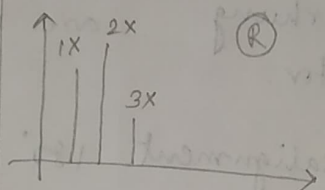
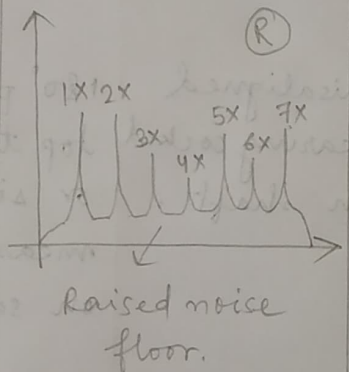
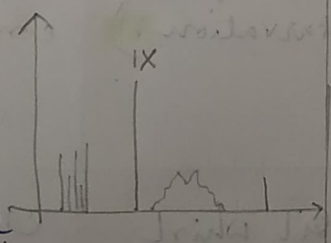
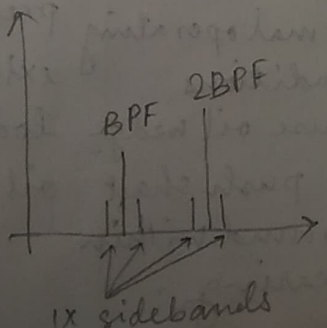
D. BELT RESONANCE

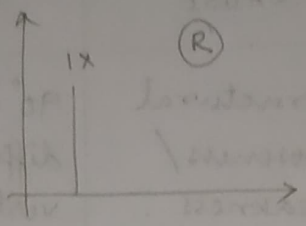
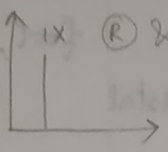
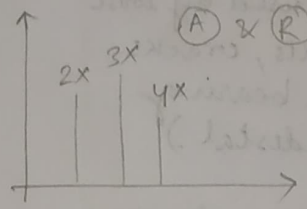
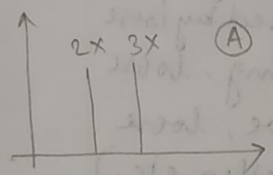
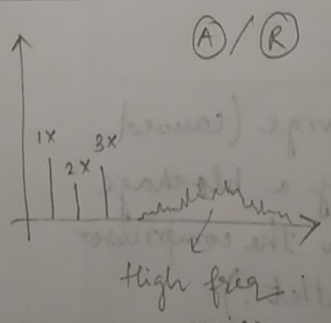
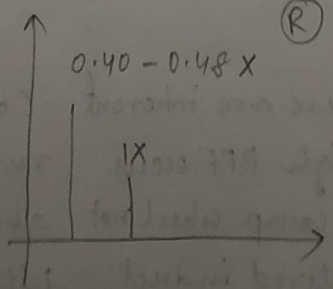




Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing belt while measuring the response on pulleys or bearings. However, when operating, belt natural frequencies will tend to be slightly higher on the tight side and lower on the slack side.

MACHINE - SPECIFIC ANALYSIS : COMPRESSORS

FREQUENCY	SPECTRUM	PROBABLE CAUSE	COMMENTS
1X (R)		Structural looseness / weakness.	90° – 180° phase difference b/w vertical measurements on bolt, machine foot, baseplate, base.
1X, 2X, 3X (R)		Bearing pedestal looseness (caused by loose bolts, cracks in bearing pedestal.)	NA
(R) 1X harmonics; subharmonic multiples of 0.5X (0.5X, 1.5X, 2.5X, etc) or 0.33X; raised noise floor.		Rotating looseness (caused by loose bearing, loose sleeve, loose impeller, etc.)	NA
vibration occur b/w 30 – 45 % of compressor speed with higher harmonics & wide skirt noise		Surge (caused by a blockage on the compressor outlet.)	NA
Blade Pass Frequency (# Blades x RPM) is high.		These are inherent. High BPF occur if comp. wheel not centered in duct (axial com.), if cut off plate is installed too close to comp. wheel (centr. comp.)	Compressor is running below/above its Best Efficiency Point.

FREQUENCY	SPECTRUM	PROBABLE CAUSE	COMMENTS
1X (R)		Unbalance (force unbalance /couple " / Dynamic ")	NA
1X (A) & (R)		Unbalance in overhung rotor.	(A) phase readings are in phase.
2X, 3X, 4X (A) & (R)		Misalignment.	180° phase diff across coupling in (A) & (R).
2X, 3X (A)		Misaligned bearing cocked on shaft.	180° phase diff from top to bottom and/or side to side as measured in (A) on same bearing.
1X harmonics may be present; high freq noise.		Cavitation (caused by insufficient inlet flow i.e. starvation.)	sounds as if stones are passing through the compressor.
0.40 – 0.48 X (R).		Oil whirl (Deviations in normal operating conditions cause oil wedge to push shaft around within bearing.	Change in oil viscosity, lube pressure and external pre loads can affect oil whirl.

COUPLING PROBLEMS

1. For Grid coupling, we will get peak at 4X.
2. For Spider coupling, we will get peak at (# jaws X RPM).
3. For Bush coupling, we will get peak at (# Bush X RPM).
4. For Metaflex coupling, we will get peak at 1X, 2X, 3X, 4X.
   ↓
   predominant.

Operating Speed Defects

Fault	Spectral Frequency	Page #	Spectrum, Time Waveform, Shape	Correction
critical speeds	1x, 2x, 3x, etc.	4.8	amplified vibration due to proximity of operating speed to natural frequency	tune natural frequency or change operating speed
mass unbalance	1x	4.9	distinct 1x with much lower values of 2x, 3x; vertical/horizontal 90° out of phase	field or shop balancing
misalignment	1x, 2x, occasionally 3x	4.10	distinct 1x with equal or higher values of 2x, 3x; vertical/horizontal in phase	perform hot and/or cold alignment
shaft bow	1x	4.12	dropout of vibration around critical speed in Bodé plot; 1x constant with speed	heating or peening to straighten rotor (allow rotor to float axially)
fluid-film bearing wear and excessive clearance	1x, sub-harmonics, orders	4.13	high 1x, 1/2x, sometimes 1/2 or orders; cannot be balanced	replace bearing
resonance	1x, 2x, 3x, etc.	4.5	high balance sensitivity, high-amplitude vibration at order of operating speed	change structural natural frequency
looseness	1x plus large number of orders, 1/2x may occur	4.13	high 1x with lower-level orders, large 1/2 order, low axial vibration	shim and tighten bolts to obtain rigidity
eccentricity	1x	4.14	high 1x	machine journal for concentricity
coupling lockup	1x, 2x, 3x, etc.	4.12	1x with high 2x similar to misalignment; start and stops may yield different vibration patterns	replace coupling or remove sludge
thermal variability	1x		1x has varying phase angle and amplitude with load	compromise balance
distortion	1x and orders		1x from preload of bearings, 2x line frequency, air gap on motor	relieve soft foot
impact-excited natural frequency	Nx	4.5	excitation of natural frequency at multiple of operating speed	remove source of pulses or impacts
sheave eccentricity	1x		spectral content depends on pitch line profile. horizontal/vertical in phase	replace sheave

Rolling Element Bearing Defects

Fault	Spectral Frequency	Time Waveform/ Spectrum Shape	Comment	Page #
outer race defect	BPFO and multiples	multiples of BPFO with sidebands of RPS, FTF, or BSF	remaining life sensitive to speed and load	4.4
inner race defect	BPFI and multiples	multiples of BPFI with sidebands of RPS, FTF, or BSF	remaining life sensitive to speed and load	4.18
cage defect	FTF and multiples	FTF and multiples or sidebands on BPFO or BPFI	worst defect; failure imminent	4.19
ball defect	BSF or FTF and multiples	may be sidebands of BPFO or BPFI	limited time to failure	4.19
excessive internal clearance	natural frequencies	natural frequencies modulated by RPS	excessive clearance from wear	4.19

Gearbox Faults

Fault	Frequency	Page #	Spectrum Time Waveform
eccentric gears	gear mesh	4.23	gear mesh with sidebands at frequency of eccentric gear
gear-mesh wear	gear mesh and harmonics	4.3	gear mesh with sidebands at frequency of worn, scored, or pitted gear(s); sometimes 1/2, 1/3, 1/4 harmonics of gear mesh
improper backlash of end float	gear mesh and harmonics	4.5	gear mesh with orders and sidebands at frequency of pinion or gear
broken, cracked, or chipped gear teeth	natural frequencies	4.23	pulses in time waveform; natural frequencies in spectrum
gearbox distortion	gear mesh and/or natural frequencies	4.24	gear mesh and orders in spectrum; varying gear-mesh amplitude in time waveform – shaft frequency plus low-amplitude orders
common factor wear	fractional gear mesh and/or natural frequencies	4.3	fractional gear mesh frequencies dependent on the common factor(s)

Pumps and Piping

Fault	Frequency	Time Waveform/ Spectrum	Correction
structural resonance/piping and vertical pump	1x or vane pass	vibration focused at RPM or vane pass	change natural frequency/speed
acoustical piping resonance	vane pass	vibration focused at vane pass	change piping length or install damper
impeller eccentricity	1x or vane pass	vibration focused at 1x RPM	install new impellers
excessive impeller-diffuser clearance	1x, vane pass	vane-passing frequencies and multiples	change clearances
recirculation	random noise	random noise on data	change operation to best efficiency point (BEP)
cavitation	random noise, vane pass	random noise on data	change operation to BEP
excessive wear ring clearance	1x	strong 1x	may alter critical speed, damping, and stability

Fans

Fault	Frequency	Time Waveform/ Spectrum	Correction
structural resonance	1x, blade pass (BP)	operating speed/ BP	change isolators
structural cracks in impeller wheel	1x	nonrepeatable phase on balancing	repair wheel
loose wheel on shaft	1x and multiples	erratic phase	repair shaft
aerodynamic problems	1x, BP	random noise and vane pass	change damper position or ducting
pedestal stiffness, asymmetry	1x	large difference in horizontal-to-vertical amplitude	stiffen pedestal
belts and pulleys	1x, 2x, belt frequency	belt pulses, 1x vertical-horizontal in phase	replace pulley or belt
acoustical duct resonance	1x, vane pass	vibration focused at one fan speed	change ducting or fan speed

Motor Malfunctions — Electrical Faults

Fault	Frequency	Page #	Spectrum; Time Waveform/Orbit	Correction/Comment
air-gap variation	120 Hz	4.28	120 Hz plus sidebands, beating 2x with 120 Hz	center armature relieving distortion on frame; eliminate excessive bearing clearance and/or any other condition that causes rotor to be off center with stator
broken rotor bars	1x	4.28	1x and sidebands equal to (number of poles x slip frequency)	replace loose or broken rotor bars
eccentric rotor	1x	4.29	1x, 2x/120-Hz beats possible	may cause air-gap variation
stator flexibility	120 Hz		2x/120-Hz beats	stiffen stator structure
off magnetic center	1x, 2x, 2x	4.29	impacting in axial direction	remove source of axial constraint-bearing thrust, coupling
stator shorts	120 Hz and harmonics	4.30	120 Hz and harmonics	replace stator

Spectrum Analysis Table

	Primary Plane	Detection Units	Dominant Frequencies	Phase Relationship	Comments
Imbalance				**• NOTE: Phase references are accurate within ± 30 degrees.**	
Mass	Radial	Acceleration/ Velocity/ Displacement	1X	90 degree phase shift as sensor is moved from horizontal to vertical position. No radial phase shift across the machine or coupling.	
Overhung Mass	Axial and Radial	Acceleration/ Velocity/ Displacement	1X	Axial reading will be in phase.	Account for change in sensor orientation when making axial measurements.
Bent Shaft	Axial and Radial	Acceleration/ Velocity/ Displacement	1X	180 degree phase shift in the axial direction across the machine with no phase shift in the radial direction.	
Misalignment					
Angular	Axial	Acceleration/ Velocity/ Displacement	1X, 2X	A phase shift of 180 degrees in the axial direction will exist across the coupling.	With severe misalignment, the spectrum may contain multiple harmonics from 3X to 10X. If vibration amplitude in the horizontal plane is increased 2 or 3 times, then misalignment is again indicated. (Account for change in sensor orientation when making axial measurements.)
Parallel	Radial	Acceleration/ Velocity/ Displacement	1X, 2X	A phase shift of 180 degrees in the radial direction will exist across the coupling. Sensor will show 0° or 180 degrees phase shift as it is moved from horizontal to vertical position on the same bearing.	
Combination of Angular and Parallel	Axial and Radial	Acceleration/ Velocity/ Displacement	1X, 2X	A phase shift of 180 degrees in the radial and axial direction will exist across the coupling.	
Mechanical Looseness					
Structural	Radial	Acceleration/ Velocity/ Displacement	1X	Phase shifts of 180 degrees will exist between the machine's feet, baseplate, and/or foundation if the machine is rocking.	Usually caused when the machine's foundation degrades to such an extent that it is no longer stiff, causing the machine to "rock".
Soft Foot	Radial	Acceleration/ Velocity/ Displacement	1X, 2X, …	Phase will shift when the machine foot is tightened.	Result of the machine footing coming loose from the foundation.
Wear/ Fitting	Axial and Radial	Acceleration/ Velocity/ Displacement	1X, 2X, … 10X	Phase reading will be unstable from one reading to the next.	Vibration amplitudes may vary significantly as the sensor is placed at different locations around the bearing. (Account for change in sensor orientation when making axial measurements).

Spectrum Analysis Table

	Primary Plane	Detection Units	Dominant Frequencies	Phase Relationship	Comments
Local Bearing Defects				• **NOTE: Phase references are accurate within ± 30 degrees.**	
Race Defect	Radial	Acceleration/ Enveloping/ *SEE*	4X … 15X	No correlation.	With Acceleration measurements, bearing defect frequencies appear as a wide "bump" in the spectrum. Bearing defect frequencies are non-integer multiples of running speed (i.e., 4.32 X Running Speed).
Gear Defect					
Gear Mesh	Radial	Acceleration/ Enveloping/ *SEE*	20X … 200X	No correlation.	The exact frequency relates to the number of teeth each gear has times the shaft rotation speed (running speed).
Electrically Induced					
AC Motors	Radial	Acceleration/ Velocity/ Displacement	Line Frequency (100 or 120 Hz)	No correlation.	Defect frequencies can be seen at exactly twice the line frequency.
DC Motors	Radial	Acceleration/ Velocity/ Displacement	SCR Frequency	No correlation.	DC motor problems due to broken field windings, bad SCR's or loose connections are reflected as higher amplitudes at the SCR frequencies ($6xF_L$).

IF THE DOMINANT FREQUENCY IS 1xRPM IN THE AXIAL PLANE		
Phase	Probable Cause(s)	Comments
180° out	Angular Misalignment	Suspect if a 2nd order peak exists on the coupling side bearing (on either the drive or driven machine) that is greater than .25 times the 1st order AND the axial 1st order on the same bearing is greater than .1 in./sec.
	Anti-friction bearing misalignment	Suspect if 2nd or 3rd orders are as high or higher than the 1st order AND the axial 1st order on both bearings is greater than .12 in./sec.
	Bent shaft	Suspect if 1st order on both shaft bearings is greater than .2 in./sec. AND the axial vibration on both bearings is greater than .12 in./sec.
0° (in phase)	Sheave face misalignment	See **Power Transmission Components** chapter in Volume II
	Thrust bearing failure (late stages)	Suspect if harmonic groups exist and the first component of any group is greater than 3 and less than 9 times running speed AND is not an integer multiple of the run speed

IF THE DOMINANT FREQUENCY IS 1xRPM IN THE AXIAL PLANE		
Phase	**Probable Cause(s)**	**Comments**
N/A	Helical gear imbalance	See **Component Analysis: Gears** chapter in Volume II
N/A	Helical gear misalignment	See **Component Analysis: Gears** chapter in Volume II
Any angle	Overhung rotor imbalance	Suspect if 1st order peak is greater than .12 in./sec. AND the horizontal to vertical ratio is greater than .25 and less than 4 on both bearings Note: an axial vibration of less than .1 in./sec. on both bearings may also be present at run speed

OTHER DOMINANT FREQUENCIES IN THE AXIAL PLANE			
Dominant Frequency	**Phase**	**Probable Cause(s)**	**Comments**
2xRPM	180° out axially	Shaft bent at coupling	
Bearing frequency (# balls x RPM)	N/A	Thrust bearing failure (early stages)	Suspect if harmonic groups exist and the first component of any group is greater than 3 and less than 9 times running speed AND is not an integer multiple of the run speed
Gear mesh frequency (# teeth x RPM)		Transmission error in helical gears	Suspect if harmonics exist at ½ tooth frequency and other half-orders
		Pitch line runout in helical gears	Suspect if sidebands exist at (# teeth ± 1) x RPM and harmonics
		Faulty helical gear tooth	Sidebands may also be present
1xRPM and 2xRPM	Both axial and radial 180° out of phase	Combination of offset and angular misalignment	Suspect if a 2nd order exists on the coupling side bearing that is greater than .25 times the 1st order AND the axial 1st order on the same bearing is greater than .1 in./sec.

IF THE DOMINANT FREQUENCY IS 1xRPM IN THE RADIAL PLANE		
Phase	**Probable Cause(s)**	**Comments**
0° (in phase)	Eccentric or imbalanced sheave	See **Power Transmission Components** chapter in Volume II
	Static imbalance	Suspect if 1st order peak is greater than .12 in./sec. AND the horizontal to vertical ratio is greater than .25 and less than 4 on both bearings Note: an axial vibration of less than .1 in./sec. on both bearings may also be present at run speed
180°	Couple imbalance	
Any angle	Dynamic imbalance	
N/A	Spur gear imbalance	1xgear rotational frequency + GMF (gear mesh frequency) with 1xRPM sidebands
	Spur gear misalignment	High GMF + GMF harmonics (1xGMF, 2xGMF, 3xGMF)
	Bearing failure (late stages)	Check for bearing fault frequency (the bearing fault frequency causing the fault will be dominant)
Varies	Looseness	Suspect loose anti-friction bearings if ½ orders are present Suspect mechanical looseness if 1st order peak is greater than .1 in./sec. AND a harmonic group of run speed exists If you are analyzing a pump, suspect loose rotating impellers if phase varies from startup to startup
N/A	Broken rotor bar	Number of poles times slip frequency sidebands will be present around 1xRPM and 1 times line frequency, 5 times line frequency, 7 times line frequency Sidebands are **not** present when motor is unloaded ("running solo")
	Eccentric rotor	Number of poles times slip frequency sidebands will be present around 1xRPM, 1 times line frequency, 2 times line frequency, 7 times line frequency Sidebands **are** present when motor is unloaded ("running solo")
	Vertical resonance	Suspect if 1st order peak is greater than .15 in./sec. and the horizontal/vertical ratio is less than .25
	Horizontal resonance	Suspect if 1st order peak is greater than .15 in./sec. and the horizontal/vertical ratio is greater than 4

OTHER DOMINANT FREQUENCIES IN THE RADIAL PLANE			
Dominant Frequency	**Phase**	**Probable Cause(s)**	**Comments**
1 x RPM and 2xRPM	Both axial and radial 180° out of phase	Combination of offset and angular misalignment	Suspect if a 2nd order exists on the coupling side bearing that is greater than .25 times the 1st order AND the axial 1st order on the same bearing is greater than .1 in./sec.
2xRPM	180°	Offset misalignment	
		Loose sleeve bearings	Other multiples of RPM may be present
$\frac{1}{2}$ x RPM	N/A	Rotor rub	Suspect if sleeve bearings are used and ½ orders are present
Sub-harmonics of RPM	N/A	Oil whirl	Suspect if sleeve bearings are present and 1st, 2nd, or 3rd order frequency divided by run speed is less than .49 and greater than .3
	Unstable	Oil whip	Suspect oil whip if sleeve bearings are present and shaft frequency is above the second critical
	Varies	Sleeve bearing failure (early stages)	
Random broadband	N/A	Cavitation	This condition is called **starvation** on fans Harmonics of running speed and broadband noise may also be present
Bearing frequency (# balls x RPM)		Bearing failure (early stages)	Check for early presence of bearing fault frequencies
2 x belt frequency		Mismatched, worn, or stretched belts	Vibration will be inline with belt Adjustable sheaves may also vibrate at 2 x belt frequency
Gear mesh frequency (# teeth x RPM)		Transmission error in spur gears	Suspect if harmonics exist at ½ tooth frequency and other half-orders
		Pitch line runout in spur gears	Suspect if sidebands exist at (# teeth ± 1) x RPM and harmonics
		Faulty tooth in spur gear	Sidebands may also be present
# blades x RPM		Blade/vane pass	
2x line frequency		Loose windings	Sidebands will be present around 2 x line frequency and around slot frequency
		Loose iron	

OTHER DOMINANT FREQUENCIES IN THE RADIAL PLANE			
Dominant Frequency	**Phase**	**Probable Cause(s)**	**Comments**
2 x line frequency	N/A	Stator problems	
		Loose stator	
		Phase imbalance	
		Stator out of round	